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J. H. Kim
DAEWOO Electronics Co.

C. W. Gu
DAEWOO Electronics Co.

S. Y. Park
DAEWOO Electronics Co.

D. W. Yun
DAEWOO Electronics Co.

Y. H. Choi
DAEWOO Electronics Co.

DESIGN OF A HIGH EFFICIENCY SCROLL WRAP PROFILE FOR ALTERNATIVE REFRIGERANT R410A

Jae-ho Kim, Chi-wook Gu, Sung-yeon Park, Duke-won Yun, Young-hoon Choi

DAEWOO Electronics Co., LTD.
Air Conditioner Business Unit
515-1, Hyosung-dong, Kyeyang-ku, Inchon, Korea

ABSTRACT

The scroll compressor, using R410A that has higher density and works at higher pressure than R22, is required to redesign its geometry such as wrap profile. The design parameters of wrap profile such as base circle radius, wrap height, involute end angle and discharge port have to be well determined to obtain high compressor efficiency. At first, a simulation program has been developed to predict scroll compressor performance. Using the program the optimized wrap profiles were designed by parametric study. Later, deformation of the wrap profile was examined by FEM analysis.

NOMENCLATURE

a	: Base circle radius	h	: Wrap height
t	: Wrap thickness	V_s	: Stroke volume
N	: Scroll turns	ϕ_s	: Outer involute start angle
ϕ_e	: Involute end angle	θ_d	: Discharge start angle
P_s	: Suction pressure	P_d	: Discharge pressure

INTRODUCTION

R410A works at higher pressure and performs cooling capacity as much as R22 with smaller stroke volume. Compressor performance decreases when the R410A is used in the conventional R22 compressor without geometry change, because increased gas force causes higher mechanical loss, and optimum geometry must be changed. The most important design parameters are base circle radius a, wrap height h, wrap thickness t, involute end angle ϕ_e, and discharge start angle θ_d. In the many past studies such as Ishii et al. had dealt with parameters a and h. However it is difficult to find studies dealt ϕ_e, which we found affecting compressor efficiency noticeably. In this study, we calculated performances with various design parameter of scroll compressor using R410A and designed optimized scroll wrap profile. Then, we examined its deformation characteristics with thermal and pressure load using FEM analysis.

PARAMETRIC STUDY

If a cooling capacity is selected, stroke volume is determined assuming volumetric efficiency. Since stroke volume is represented as $V_s = 2\pi a h(\pi a - t)(2\phi_e - 3\pi)$, base circle a becomes;

$$a = \frac{t + \sqrt{t^2 + 2V_s / h(2\phi_e - 3\pi)}}{2\pi} \tag{1}$$

Decision of a discharge start angle θ_d is important to minimize discharge loss caused by over and under compression. Optimum θ_d is related to the shape and area of discharge port, and inner end of scroll wrap. If the pressure in the chamber that is about to discharge is determined, to make discharge flow smoothly, the inner end of the orbiting scroll wrap must leaves the inner involute wrap of fixed scroll when the compression chamber is just connected to discharge port. Therefore we are able to get the ϕ_s from the θ_d with geometrical ralation as follows Eq.(2).

$$\phi_s = \phi_e - 2\pi N - \theta_d - \pi \tag{2}$$

It is not easy to determine wrap thickness t, because it is related to not only compressor performance but also wrap strength. The thicker t, the less leakage and the stronger wrap, but the heavier orbiting scroll and the more friction loss at journal bearing. Therefor the t has to be selected adequately.

CALCULATION RESULT

Compressor performance is calculated at the operating condition T_{eva}=7.2°C, T_{cond}=54.4°C, P_s=995kPa, P_d=3376kPa, and $T_{suction}$=18.3°C. Stroke volume is fixed to V_s=17.5cc from cooling capacity, and 3mm is selected for wrap thickness.

From Eq.(1), relation between a and h about each ϕ_e 900°, 990°, 1080°, 1170°, and 1260° is shown in Fig.1. The pressure with respect to crank angle is shown in Fig.2, for each wrap profile marked as • on Fig.1. For to specifically evaluate gas flow of discharge process, it is required to calculate discharge port area. However, it is difficult to calculate discharge port area for every case of wrap profiles. In our parametric study, we assumed that the discharge starts when the pressure in the compression chamber reaches 90% of the P_d, then the discharge area increases linearly up to 50mm^2 after the crank shaft rotates 120°. If ϕ_e is small, the compression process ends in shot period of time. Therefore even though the leakage rate between compression chamber becomes slightly higher (because of rapid pressure development), the entire amount of leakage through compression process becomes smaller. Hence, the adiabatic compression efficiency increases as the ϕ_e decreases as shown in Fig.3(a). Fig.4 and Fig.5 show the tangential and the axial gas forces respectively for each case of ϕ_e. Forces acting on the moving parts are proportional to gas forces. When the ϕ_e is small, the maximum radius and area occupied by wrap is also small, accordingly the tangential and axial gas forces become small, therefore the mechanical efficiency becomes higher as shown in Fig.4 and Fig.5. The combinations of geometry from Fig.1 were put into the simulation program, and the result of EER is shown in Fig.6. The marked with • in Fig.6 represent maximum EER for each ϕ_e, and their geometry are marked with • in Fig.1. As the ϕ_e is small, the optimum h is large and the EER is inversely proportional to the ϕ_e, which is shown in Fig.7. The EER difference is up to about 3% between ϕ_e=900° and 1260° the one additional turn.

Some scroll wrap profiles out of those marked with • in Fig.1 are plotted in Fig.9. As mentioned above, the crank rotation angle is the one when the pressure in the compression chamber is reached up to 90% of Pd. The discharge ports are plotted as bolded circle at the center of fixed scroll. The area of discharge port decreases as ϕ_e decreases as shown in Fig.8. Though we did not consider the effect of discharge loss, we can predict that the loss will increase as discharge port becomes too small. Since the discharge loss affects the adiabatic efficiency, it may not always increase as the angle θ_d decreases. The dashed line in Fig.3(a) represents it.

FEM ANALYSIS

Using the results from the simulation, an FEM analysis has been done on the optimized wrap profile to evaluate deformation and reliability. The pressure and temperature at maximum operating condition were applied. As a result, the maximum axial deformation is about 0.1mm, which we must consider the initial gap setting to avoid contacting scrolls each other during operation. The resulted maximum equivalent stress is about 30% of yield stress of the cast iron. Also it is found the axial deformation is mainly dominated by temperature and the radial deformation is dominated by teperature and pressure.

CONCLUSION

1. We designed the optimum wrap profile using a developed performance simulation code.
2. We found that the smaller involute end angle ϕ_e, the better EER.
3. Optimum wrap height is proportional to ϕ_e.
4. For the optimum wrap profiles, deformation and maximum equivalent stress were assessed as within controlable range.

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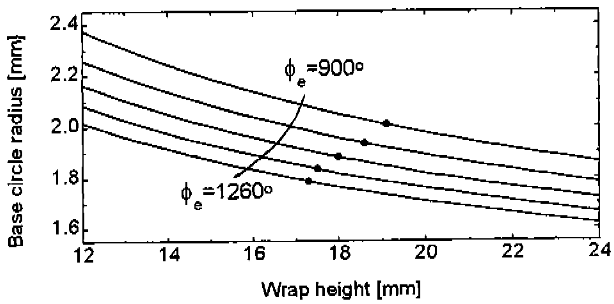


Fig.1 Wrap height Vs. base circle radius

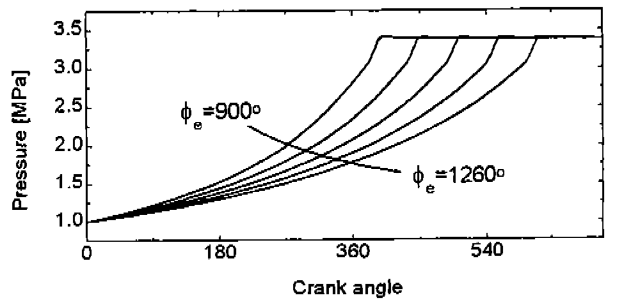


Fig.2 Pressure in compression chamber

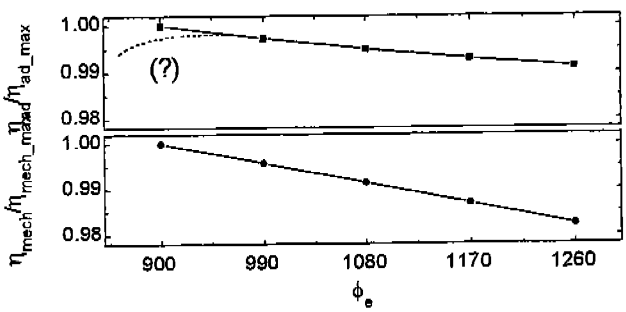


Fig.3 Efficiency : (a)Adiabatic, (b)Mechanical

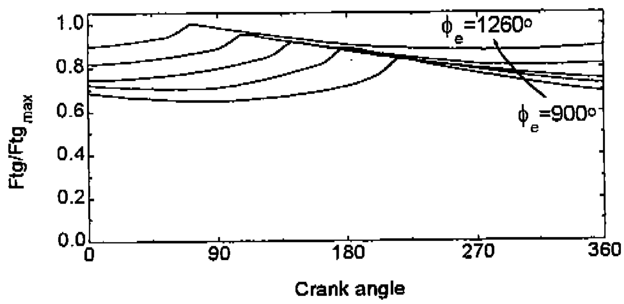


Fig.4 Tangential gas force

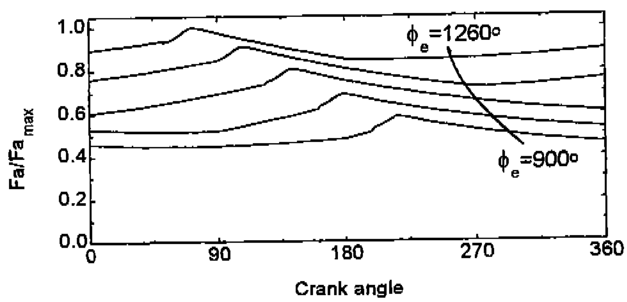


Fig.5 Axial gas force

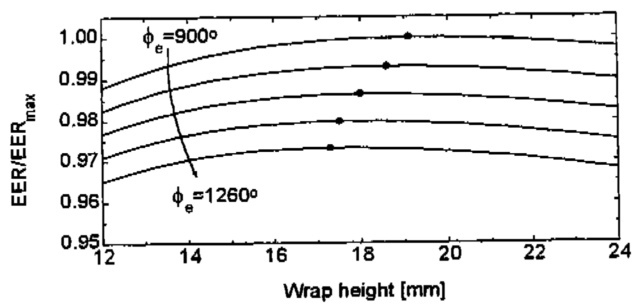


Fig.6 Maximum EER

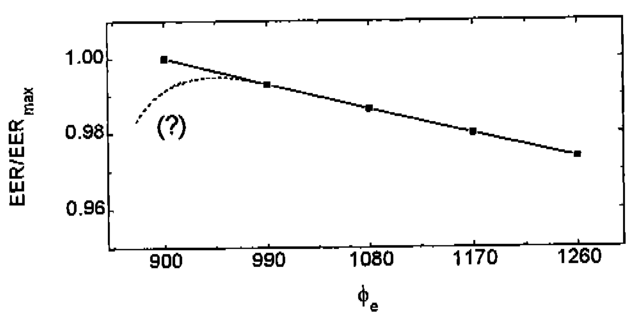


Fig.7 EER Vs. involute end angle

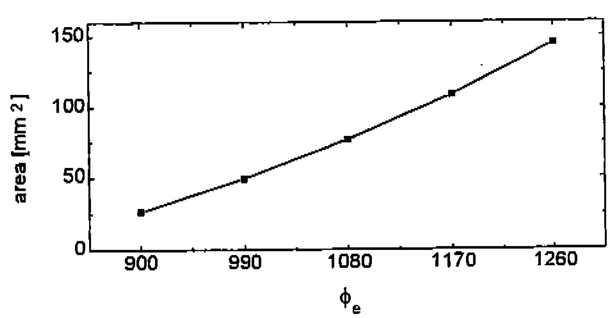
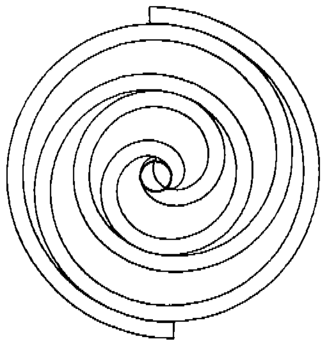
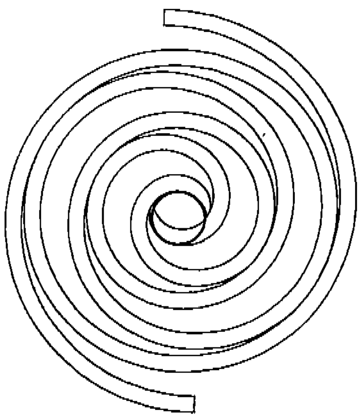


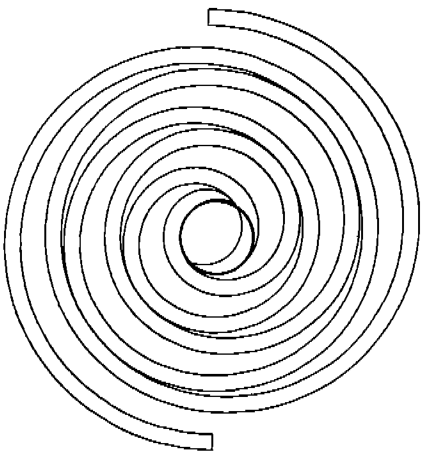
Fig.8 Discharge port area



(a) ϕ_e=900°, θ=19.1°



(b) ϕ_e=1080°, θ=124.3°



(c) ϕ_e=1260°, θ=232.9°

Fig.9 Scroll wrap and discharge port

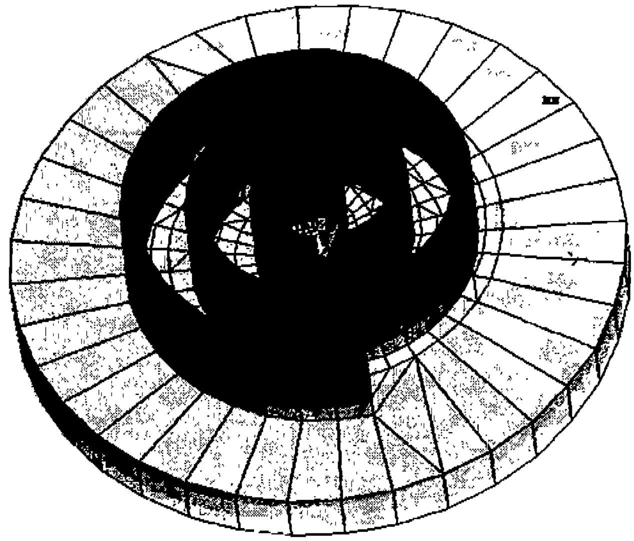


Fig.10(a) Axial deformation of orbiting scroll

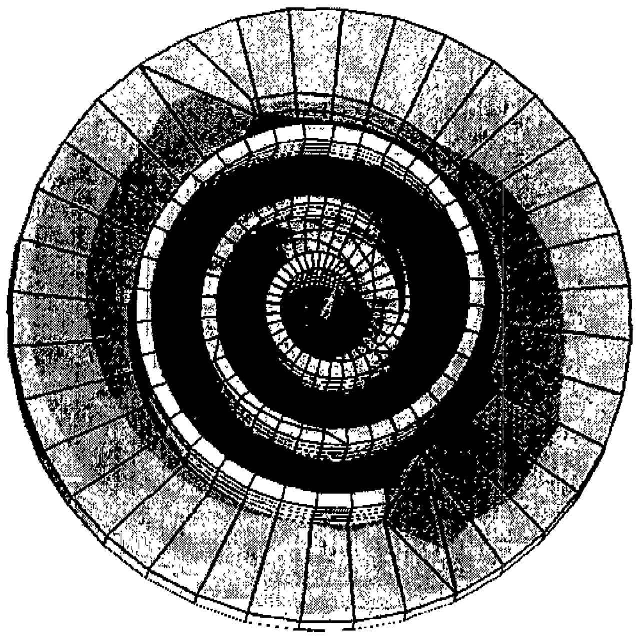


Fig.10(b) Von mises stress of orbiting scroll